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A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine

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BY MEANS of computers it is now possible to calculate the pressure and temperature trends in the diesel engine cylinder, and show close agreement with actual conditions. Such programs afford rapid and convenient provision of a wealth of interesting data, such as thermal and mechanical load and engine efficiency. It is no longer necessary to use simplified theoretical processes whose results are usually unsatisfactory. The differential equations that describe the change of state of the working fluid in the cylinder along the crank angle can now be solved step by step.

Recently, a number of publications dealing with this subject have appeared (1-4).* In studying this material, it becomes evident how imperfect is our knowledge of one of the boundary conditions that are essential for this calculation, that is, the heat transfer to the cylinder walls. This is due to the fact that the process in the cylinder not only changes locally, but also undergoes rapid changes in terms of time, so that it is very difficult to take measurements.

The experiments often provide data in a lumped form only, that is, are simultaneously dependent upon a great many parameters. Only by electronic calculation can we obtain a more precise analysis of the measured data. This explains why this problem has become increasingly important and why a number of papers have appeared recently (5-8), drawing attention to the deficiencies of existing equations and proposing new formulas for calculating heat transfer coefficients in the engine.

Because any calculation of engine events necessitates the assumption of local average values, it is essential to find an equation for the instantaneous, local mean heat transfer coefficient in the engine. Such an equation, along with mean gas temperature T, the surfaces A_i (piston, cylinder cover, and liner instantaneously participating in the heat transfer), and their relevant mean surface temperatures T_{wi}, when integrated over the crank angle should give the total heat transferred during the entire cycle, according to

$$Q_w = \int_{\text{cycle}} \sum_i h A_i (T - T_{wi})\, d\phi \qquad (1)$$

*Numbers in parentheses designate References at end of paper.

ABSTRACT

This article introduces a new method for determining the heat transfer coefficient for internal combustion engines. From the laws of similarity governing convective heat transfer, the author has derived an equation containing two convective terms, one of which takes into account the piston motion and the other the convection due to combustion. These terms, expressed as constants, relate fluid flow conditions during the scavenging periods and take into consideration the type of combustion system. The result is a universally applicable equation designed for evaluating heat transfer coefficients.

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A number of authors have already tackled this problem (9-13), and these theoretical and experimental investigations have produced various formulas for the heat transfer coefficient in the engine. However, these formulas, on being applied to a given engine in accordance with Eq. 1 give us results (as may be seen readily from Table 1) that differ as much as 200%. All the equations shown in Table 1, excepting that of Annand, are based on the Nusselt formula, found by trials on a spherical bomb:

$$h = 0.99 \sqrt[3]{p^2 T} \qquad (2)$$

In the research program described in this paper, the Nusselt experiments have been repeated, and it seems appropriate to begin with a discussion of these results because they shed light on some fundamental questions.

In a spherical bomb, as shown in Fig. 1, a combustible gas-air mixture is ignited centrally by means of a spark. Pressure and temperature in the bomb at first rise rapidly, and on completion of combustion decrease, owing to the heat transfer to the walls. The pressure can be measured by an indicator, and the time curve of the local mean gas temperature is then calculated from this pressure by applying the equation for the state of the gas. The first law of thermodynamics, in differential form and related to the time employed to the gas, is

$$\frac{dQ_c}{d\tau} = \frac{dU}{d\tau} - \frac{dQ_w}{d\tau} \qquad (3)$$

and shows that the heat generated during the reaction Q_c causes a change in the internal energy U and in the heat transferred to the wall of the bomb.

Once combustion is complete, the left-hand term of Eq. 3 becomes zero, so that it is possible from then on to determine the heat-time curve and the heat transfer coefficient directly from the change of the internal energy of the gas. This is the very method already employed by Nusselt; it fails, of course, during the combustion phase because there $dQ_c/d\tau$ is unknown and changeable in terms of time. By integrating Eq. 3, it is at least possible, however, to ascertain from the heat balance the total heat transferred to the wall during the combustion period.

The second method, which is independent of this one and frequently applied to internal combustion engines, involves measuring the time curve of the wall surface temperature, and using it as a boundary condition for solving the Fourier differential equation that describes the temperature pattern in the wall as a function of time and location. Proceeding from this, it is possible to find the temperature gradient on the surface, which is a measure of the heat transfer coefficient. Thus heat transmission can be studied on a straightforward and simple model, and the results of the two methods for determining the heat transfer coefficient can be compared with each other.

Table 1 - Heat loss to the walls, integrated from inlet closure to exhaust opening, applying different formulas for heat transfer coefficients

Proposed by		Q_w, % of Fuel Heat
Formula		
$h = 0.99(1 + 1.24 c_m)\left(\sqrt[3]{p^2 T}\right)$		
$h = 0.99(3.5 + 0.185 \times c_m)\left(\sqrt[3]{p^2 T}\right)$		
$h = 2.1\left(\sqrt[3]{c_m}\right)\left(\sqrt{pT}\right)$		
$h = f_1(p_L) \times f_2(c_m) \times \sqrt{pT}$		
$h = 0.49\left(\frac{k}{d}\right) Re^{0.7} + 0.91 \times \frac{\left[\left(\frac{T}{100}\right)^4 - \left(\frac{Tw}{100}\right)^4\right]}{T - T_w}$		
Nusselt	1923	15
Briling	1931	8
Eichelberg	1939	13
Pflaum	1962	23
Annand	1963	22

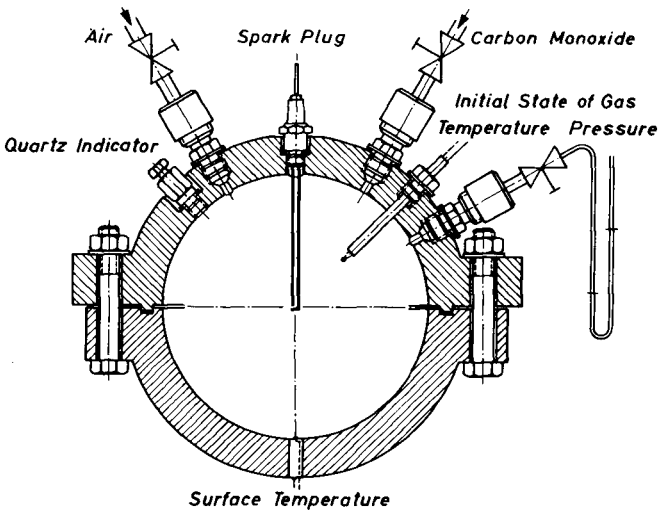

Fig. 1 - Spherical combustion chamber; arrangements of measuring points

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Fig. 2 gives an example of the evaluation of such results. The upper section of Fig. 2 gives the pressure and surface temperature versus time from which the heat transfer coefficient given in the lower section of Fig. 2 can be calculated. The fully drawn curve represents the heat transfer coefficient as calculated from the measured surface temperature h_{Tw}, and the broken curve is the heat transfer coefficient as determined from the change of the internal energy of the gas h_U. The latter can be calculated only on completion of combustion, a point that can be assessed from the time curve of the change of internal energy, also given in Fig. 2. The end of combustion occurs approximately at the moment when $dU/d\tau$ becomes zero.

The results of the two methods applied here should agree, but as may be seen, the discrepancy is considerable. This was found to be due to great local differences. The evaluation of the internal energy provides a mean heat transfer coefficient in terms of location related to the total bomb surface, whereas the evaluation of the surface temperature yields only a local value at the point where the surface thermocouple is affixed, that is, in this case, the lower pole of the spherical bomb.

Although, because of the symmetrical shape of the system under discussion, such substantial local differences were at first not expected, a glance at Fig. 3 shows that the local heat transfer coefficient along the height of the bomb increases systematically upward from the bottom. The differences amount to more than 100%. This is due to the specific time curve of combustion in the bomb.

In the center of the bomb where ignition occurs, at first a hot nucleus is formed, which expands and then rises as a result of the difference in density, pushing down the gas close to the wall. Due to the heat transmission, the gas near the wall becomes colder along its path so that the heat transfer coefficient related to the average gas temperature decreases from top to bottom. Only by forming a suitable mean value from the local heat transfer coefficients (as determined from the measurements carried out on the five points shown schematically in Fig. 3) was it possible to make the results of the two methods agree. This may be seen from Fig. 4, which again gives the pressure, the change in internal energy, the time curve of the heat transferred to the wall as calculated from the local average surface temperature, and the heat transfer coefficient assessed by both methods.

Integration of the time curve of the heat transmitted to the wall, $dQ_w/d\tau$, over 0.8 sec gives an amount of 3.088 kcal, and for the same time the heat balance gives 3.164 kcal. The difference consequently amounts to less than 3%. From this can be concluded that the method which applies the surface temperature does indeed yield reliable local values. For determining the mean heat transfer coefficient in terms of location, it is suitable, however, only if a local

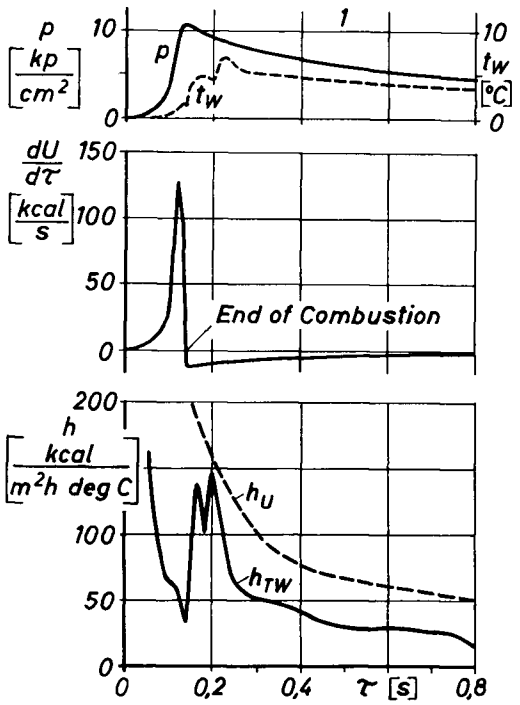

Fig. 2 - Comparison of the results of the two methods applied for determining the heat transfer coefficient; p = pressure, t_w = surface temperature, $dU/d\tau$ = change in internal energy, h_U = heat transfer coefficient calculated from internal energy, h_{Tw} = heat transfer coefficient calculated from surface temperature

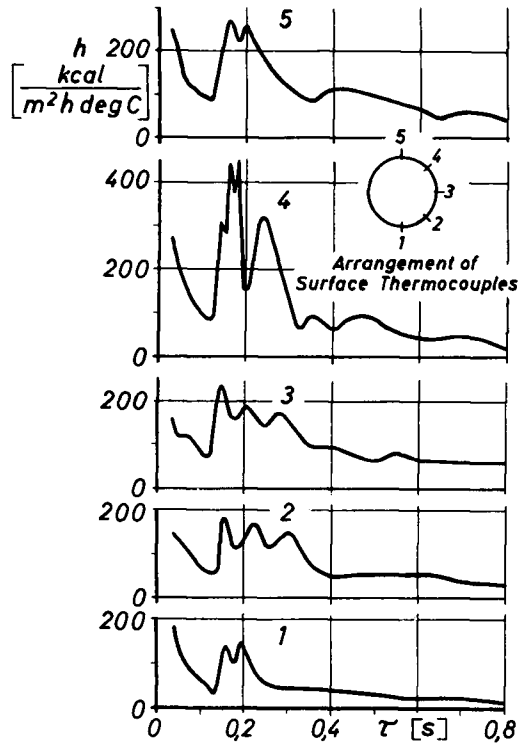

Fig. 3 - Local heat transfer coefficient calculated from the measured surface temperature

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average time curve of the surface temperature is employed. This is formed from measurements at different points of the heat-absorbing area.

Even in the straightforward case under discussion, measurements at five points were required, their local differences amounting to more than 100%. Due to the irregular geometrical shapes of their combustion chambers, substantial differences are to be expected in internal combustion engines. Consequently, to obtain a proper average value, it would be necessary to distribute a great many thermocouples over the relevant area, and this is impractical. It is therefore the view of the author that this method is unsuitable for assessing the mean heat transfer coefficient in terms of location.

The time curve of the heat transfer coefficient in the bomb as shown in Fig. 4 may be readily understood from Fig. 5, where the actual temperature pattern in the bomb, as a function of time and location, is shown schematically. As mentioned before, a hot nucleus appears at the beginning of combustion in the center of the bomb. This nucleus expands and therefore compresses the gas layer near the wall, thus raising the temperature of the latter. By reason of the heat transfer to the wall a temperature profile forms with time, so that the temperature gradient at the surface decreases rapidly. The heat transfer coefficient therefore also decreases during this first phase.

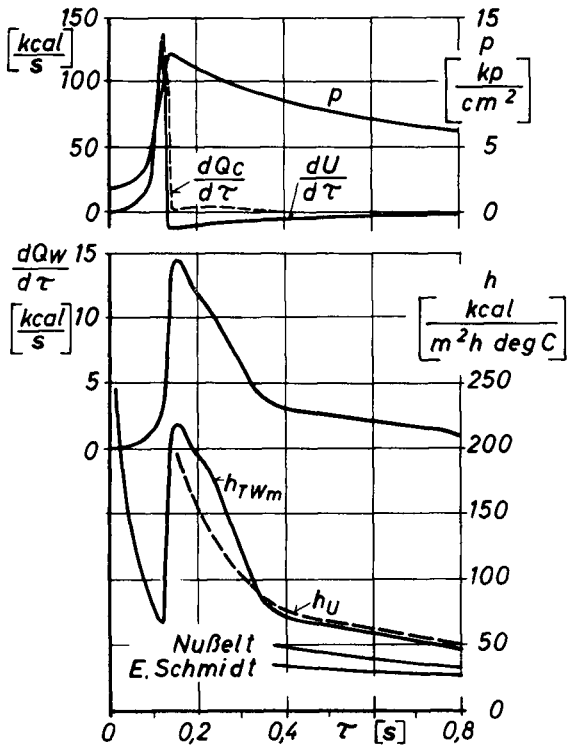

Fig. 4 - Time curves of heat release, heat loss to the wall, and local average heat transfer coefficient; $dQ_c/d\tau$ = heat release, $dQ_w/d\tau$ = heat transfer rate, h_{Twm} = local average heat transfer coefficient calculated from the five measured surface temperatures plotted in Fig. 3.

In the ensuing combustion, the hot nucleus of the gas in the center of the bomb increases and finally touches the wall, that is, the flame hits the wall so that the temperature gradient on the surface again becomes very steep and the heat transfer coefficient increases abruptly. In the following phase, heat transmission is controlled by the turbulence occurring in the flame zone when the remaining unburned gas particles close to the wall are detached and burn off slowly. This can be seen from the heat release curve, represented by the broken curve in the upper part of Fig. 4, which is provided by adding the time curve of the heat transferred to the wall, $dQ_w/d\tau$, to the trend of the change in internal energy, $dU/d\tau$. Simultaneously the aforementioned convection fluid flow is formed, which rises in the center and descends near the wall.

All these processes effected by the combustion itself rapidly die down once combustion is definitively finished, and very soon after this the heat transfer is determined by free convection only, caused and permanently maintained by the temperature differences effected by the heat transfer itself. Fig. 4 also shows the results provided by the Nusselt formula (Eq. 2). They are below the measured data. The reason for this is that Nusselt evaluated a range that was nearly 4 sec after the end of combustion, so that the influence of combustion had already faded away to a large extent.

The heat transfer caused by free convection in spherical vessels has been investigated by E. Schmidt (14). He found the equation

$$Nu = 0.65\,(Gr\,Pr)^{0.25} \qquad (4)$$

the results of which also are plotted in Fig. 4. They do not differ substantially from those found by applying the Nusselt formula. This may be taken as evidence that Eq. 2 as proposed by Nusselt does not represent the heat transfer effected by combustion but by free convection in spherical vessels.

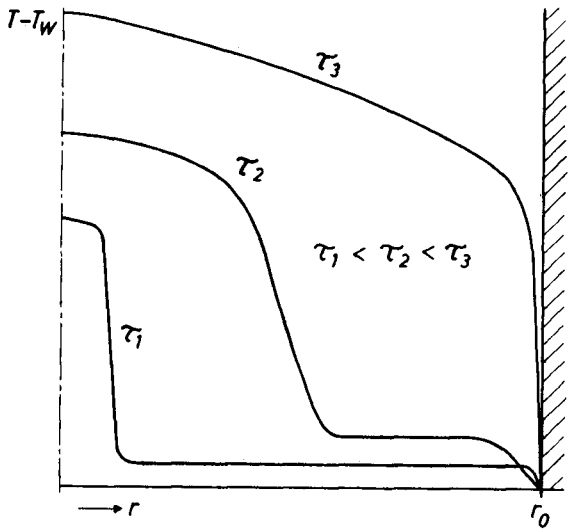

Fig. 5 - Temperature pattern in the bomb during combustion

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To his formula, Nusselt added the factor $(1 + 1.24c_m)$, to allow for the influence of piston motion in the engine. Thus the well-known Nusselt formula for the heat transfer coefficient in the engine came into being and is to this day often applied. Later, Eichelberg also assumed that the term $\sqrt[3]{p^2T}$ was suitable for representing the convective heat transfer in the engine, and changed it into $\sqrt{pT}$. In order to consider in this way the influence of radiation in a lumped form, he increased the power of T and decreased the power of "p" by empirical means.

The extension of the results investigated in the bomb, as done by Nusselt and indirectly by all other authors who have applied in their equations the term $\sqrt[3]{p^2T}$ or $\sqrt{pT}$, is inadmissible because the physical processes in both cases differ completely. In the internal combustion engine the heat transfer is largely controlled by forced convection, a requirement necessitated by the piston motion on the one hand and the combustion on the other hand. Free convection is insignificant in the engine. Therefore all equations containing the terms $\sqrt[3]{p^2T}$ or $\sqrt{pT}$ are not theoretically substantiated. They have only an empirical character and a small scope of application, which in most cases is disregarded and often not even known. That is the main reason for the deficiencies of the equations mentioned in Table 1.

The heat transfer in the engine is controlled by three important physical influences: convection, radiation, and influence of the rapid change in gas temperature as a function of time. As the gas velocity in the cylinder is high, it is here a question of forced turbulent fluid flow. In this case the convective heat transfer obeys the law

$$Nu = CRe^m \tag{5}$$

If density, viscosity, and conductivity of heat of the gas are expressed as functions of pressure and temperature, as shown by the author (8), it follows that

$$h = Cd^{m-1} p^m w^m T^{0.75-1.62m} \tag{5a}$$

where:

d = Characteristic unit of length
p and T = Pressure and temperature of the working fluid
w = Local average gas velocity in the cylinder

The gas velocity is known neither in quantity nor in its trend along the crank angle. All authors therefore apply to it the mean piston speed c_m. If one first considers the unfired engine, then the gas velocity is actually effected by the piston motion only. Therefore it is correct (if, as a first approximation, a constant average value throughout the entire cycle is taken) to write

$$w = C_1 c_m \tag{6}$$

Eq. 5 describes the convective heat transfer under steady conditions. During compression and expansion, however, the gas temperature changes rapidly with time, so that the temperature gradient during compression becomes steeper and flatter during expansion than in the steady-state case. This should effect an additional influence of the frequency of change in temperature, that is, a greater influence of the engine speed than expected from Eq. 5. Undoubtedly such an influence must occur in the engine, but its extent and the amount of the constants and exponents in Eq. 5 can be determined by experiments only. The influence of radiation in the unfired engine, because of the low temperatures, can safely be neglected.

The next step in the investigation was therefore to ascertain the heat transfer in the unfired engine. The method using surface temperature is not suitable. While it is often applied in the engine, the results published so far (15-18), indicate that, because of their considerable scatter, no information can be provided on the relationships among the heat transfer coefficient and engine speed, pressure, temperature, and fuel-air ratio. In the unfired engine, or even during the scavenging period, no reproducible values are yielded. The reasons for this are the small changes in surface temperature and, above all, the considerable local differences already mentioned above.

The only alternative is, therefore, to determine from the heat balance the heat transferred to the walls. These measurements, if carried out carefully, provide reliable values. All we obtain here, however, is the heat summed up over the whole cycle, that is, the left-hand term of Eq. 1. To be able to determine therefrom the heat transfer coefficient, it is necessary to assume first its probable trend in order to use this in calculating the total heat transferred to the walls during the whole cycle, in accordance with Eq. 1. This value is then compared with the heat value known from the heat balance.

By occasionally changing one parameter such as pressure, temperature, and engine speed, it is possible to check the relationship originally assumed. An essential condition for calculating the heat losses to the walls is a knowledge of the gas temperature in the cylinder, which is determined from the measured pressure by means of the equation of the state of the gas. For this purpose it is necessary, however, to know the value of the gas temperature at one point. This can be either assumed or, more accurately, calculated by means of computers, simulating the engine cycle and including the scavenging period (19). To this end the heat transfer conditions in the induction phase must be known.

The first step was to investigate the scavenging period. A four-stroke engine was fitted with a special camshaft with double-symmetrical cams so that, when motored, it would aspirate and expel once per revolution, compression and expansion strokes being eliminated. If hot gas is fed into this engine it can be measured quasisteadily like a heat

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exchanger. The test engine is shown in Fig. 6. The gas is heated in a combustion chamber, induced by the engine, and expelled instantly, at which time part of its heat is transferred to the inlet and exhaust throats and the cylinder walls. The following data are measured:

1. Heat given off by the gas and that absorbed by the cooling water and lubricating oil.
2. Power.
3. Pressure curve in the cylinder.
4. Time average surface temperatures.

To separate the heat transferred in the inlet and exhaust throats from the total heat in order to determine the amount actually transferred to the cylinder walls, the cylinder cover was first measured under steady-state conditions. Thus the heat transfer coefficients in the throats were investigated as a function of valve lift and Reynolds number. On applying these values, it was possible to calculate electronically by steps the heat transferred in the throats, taking the instantaneous values of valve lift and mass flow rate into consideration. By subtracting this calculated amount from the total heat measured, it was then possible to determine the heat actually transferred to the cylinder walls during the scavenging period.

By introducing Eq. 6 into Eq. 5a and by arbitrarily employing first the exponent 0.8, which applies to turbulent fluid flow in pipes, one finds (8)

$$h = 110\left[d^{-0.2} p^{0.8} (C_1 c_m)^{0.8} T^{-0.53}\right] \qquad (5b)$$

From this equation the heat transfer coefficient and, in accordance with Eq. 1, the total heat transferred to the cylinder walls during the scavenging period was calculated. The result compared with the measured data, and thus by changing successively one parameter (that is, pressure, temperature, and mean piston speed), it became possible to check the originally assumed relationship. Fig. 7 gives the results of this investigation and shows that the assumed relationship in accordance with Eq. 5b is fully corroborated. In the special case (that is, two inlet valves without any swirl), the constant C_1 was determined as being 6.18.

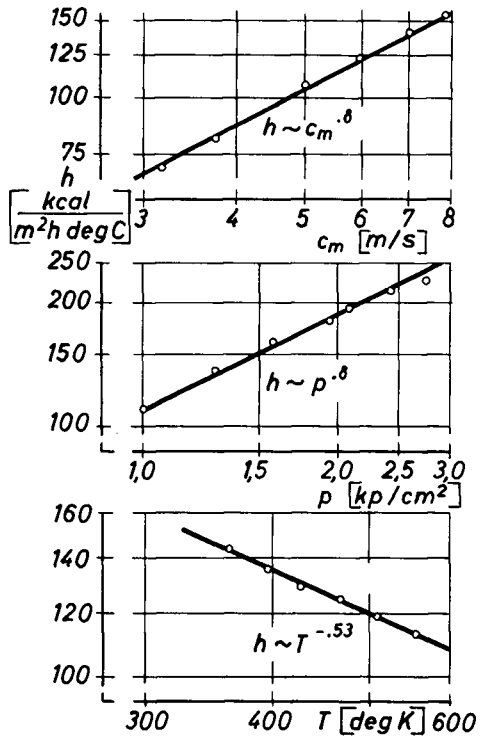

Fig. 7 - Heat transfer coefficient during scavenging period

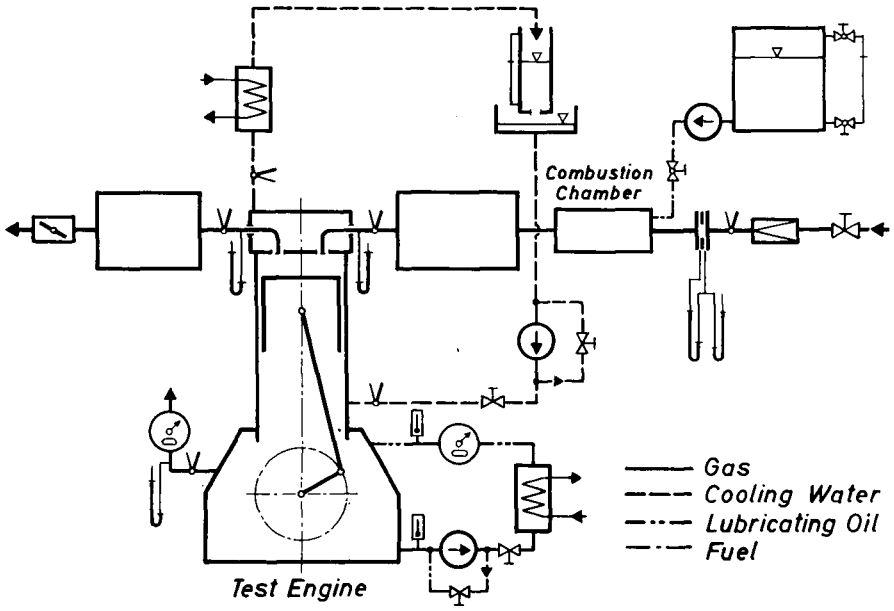

Fig. 6 - Scheme of test equipment

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The heat instantaneous transfer coefficient during the scavenging period can be calculated by Eq. 5b. Since the constant C_1, the proportional factor between the mean piston speed and the unknown actual mean gas velocity, depends upon the fluid flow conditions during the induction phase, Eq. 5b is universally applicable only in the case of engines whose cylinder cover configurations are geometrically similar. Different cylinder cover designs necessitate investigations of other constants on an experimental unit that is representative of a given engine group. These trials relating to the scavenging period have been carried out by Zapf (20).

Once the heat transfer conditions during the scavenging period were known, the next step was to determine in the same manner the total heat transferred during the entire cycle, including the compression and expansion strokes, but still without combustion. The same engine, this time equipped with a normal camshaft, was motored and fed with hot gas.

By subtracting from the total heat measured the amount of heat transferred to the throat and cylinder walls during the scavenging period (this can be calculated by means of the already known heat transfer coefficients), only the amount of heat transferred to the cylinder walls during the compression and expansion strokes can now be calculated. If this amount of heat is again calculated by means of Eqs. 1 and 5b, and compared with the measured data, another constant C_1 is found for the compression and expansion phase.

The gas motion generated during the induction phase dies away with time, owing to internal friction. In the compression stroke, however, dependent upon the piston or cylinder cover shapes, an additional gas motion (squash swirl) can be induced, whose intensity is also proportional to the mean piston speed. In this particular case, the engine involved had direct injection and straightforward combustion chamber walls, so that the decay due to internal friction prevailed. The mean gas velocity during the time the valves are closed is considerably smaller than during the scavenging period; the constant C_1 applicable to the compression and expansion phase was therefore found to be 2.28 only. This, of course, again depends upon engine design.

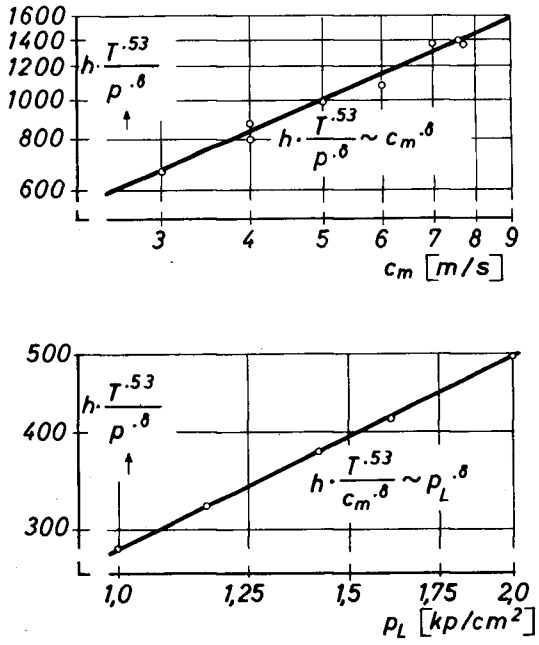

Fig. 8 - Related heat transfer coefficient during compression and expansion strokes

In the phase under discussion, the aforementioned influence of the fluctuating change in gas temperature should occur and result in a greater exponent for the mean piston speed. In order to prove this, two test series were undertaken, changing first the pressure and then only the engine speed. From Fig. 8 it may be seen that the exponents of both parameters are equal, as required by the theorem of similarity of steady-state convective heat transfer. From this result we may conclude that the influence under discussion is negligible on the engine, at least in the speed range $400 < n < 900$ rpm studied here. Nusselt was of this opinion over 30 years ago (21). However, a number of authors (15, 17, 22, 23) consider this influence to be of prime importance. To this author's knowledge, however, experiments have failed so far to demonstrate this influence on the engine.

The heat transfer in the unfired engine can be calculated with adequate certainty by Eq. 5b, derived from the theorem of similarity of steady-state convective heat transfer. For the unknown gas velocity, the respective average values in terms of time relating to the scavenging period and the compression and expansion phase are applied.

On the same engine, but this time fired, the total heat transferred to the walls during the entire cycle was determined from the measured heat balance. Fig. 9 shows an example for such a measurement. The heat loss to the walls is computed as the difference between the input fuel heat Q_C and the sum of indicated work Q_i and exhaust gas energy Q_{EX}. The remainder, Q_R, amounts to about 10% on single cylinder engines because of the considerable heat losses to the surroundings, and is included, whereas the friction heat Q_F is not taken into consideration.

If the heat loss to the walls is calculated by Eqs. 1 and 5b, the computation provides less than the measured amount. In the lower part of Fig. 9 this calculated heat is plotted, separated into the various terms. During the scavenging period, Q_{TH} is absorbed by the throat and Q_{SC} by the cylinder walls. Q_{COMP} is transferred during compression and Q_E during the expansion phase. Since the physical conditions in the unfired and the fired engine are largely equal during the scavenging and compression phase, the amounts calculated from Eqs. 1 and 5b for this period can be assumed to be in agreement with the actual conditions. The surplus seen in Fig. 9 must therefore be transferred during the combustion and the expansion phase only. This is effected by

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the additional influence of combustion occurring in the combustion period and decaying in the subsequent expansion stroke.

The essential physical reasons for influence are the turbulence caused by the combustion itself on the one hand and the radiation on the other hand. To be able to investigate this combustion influence separate from the engine, the experiments on the bomb, discussed above, were carried out. To this end an attempt should be made to interpret the time curve of the heat transfer coefficient during the combustion phase (as plotted in Fig. 5) by means of a suitable equation. But it is during this very phase that the heat transfer is substantially influenced by the specific time curve of combustion in the bomb.

In diesel engines a jet of fuel is injected and each droplet burns off simultaneously, distributed more or less uniformly over the whole combustion chamber so that differences in terms of location and time have an entirely accidental character. A systematic flow of combustion from the inside outward and a sudden simultaneous penetration of the flame toward the wall as it occurs in the bomb can therefore not be expected in diesel engines. For this reason there is little use in extending to the diesel engine the results investigated in the bomb.

Determination of the combustion influence is therefore possible only in the engine itself. The sum of the heat transferred during the combustion and expansion period and its dependence upon the main parameters (such as the state of aspirated air, engine speed, and input fuel rate) can be investigated experimentally as explained in Fig. 9. The main difficulty consists in finding a simple expression that gives this additional influence not only as a function of the input fuel rate, but also in terms of time. If one considers first the additional gas motion effected by combustion, for which we may introduce an additional gas velocity w_c, and which in the combustion and expansion phase is superimposed on the velocity caused by the piston motion, one can write

$$w = C_1 c_m + w_c \qquad (6a)$$

From the time curve of w_c we only know that it increases rapidly from zero in the beginning of combustion, reaching

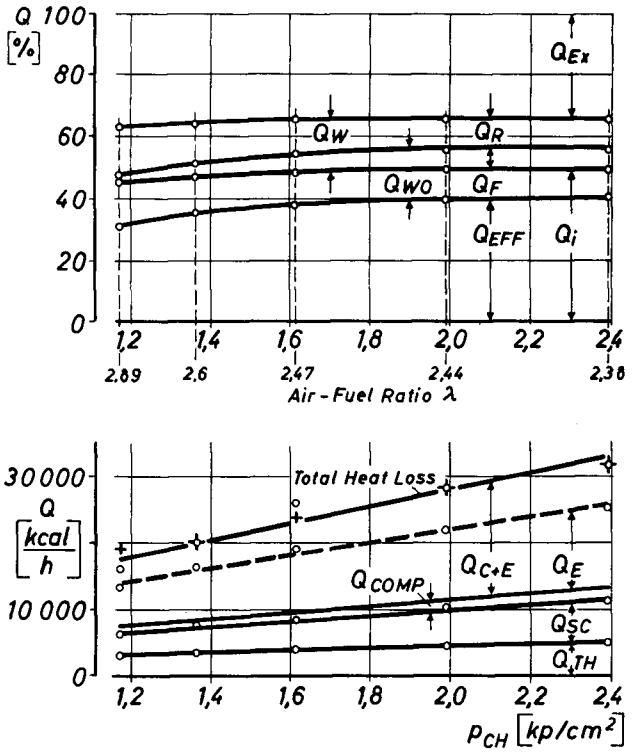

Fig. 9 - Heat balance of the fired engine, c_m = constant; Q_{EFF} = effective power, Q_i = indicated power, Q_{Ex} = exhaust energy, Q_{WO} = heat absorbed by water and lubricating oil, Q_F = friction energy, Q_R = remainder, Q_W = total heat transferred to the engine, Q_{TH} = heat transferred to the throat walls, Q_{SC} = heat transferred to the cylinder walls during scavenging period (Eq. 5b), Q_{COMP} = heat transferred during compression stroke (Eq. 5b), Q_E = heat transferred during expansion phase (Eq. 5b), Q_{C+E} = heat transferred during combustion and expansion phase (+ points calculated by Eq. 8)

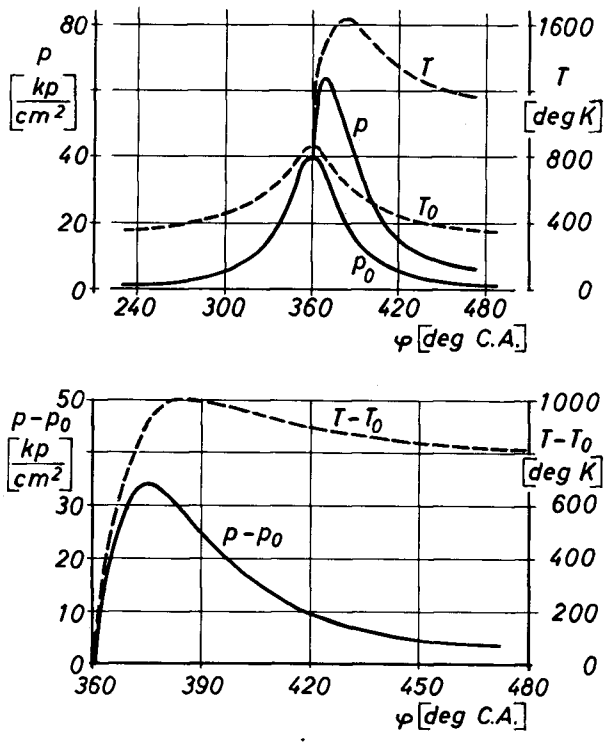

Fig. 10 - Change of temperature $T - T_0$ and pressure $p - p_0$ in the cylinder caused by combustion

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a maximum and then decaying as a result of internal friction in the expansion phase. The greater the input fuel rate, the greater the absolute height of the curve. The heat release pattern has approximately this time curve. Unfortunately it is not directly known but must be determined by complex calculations from the measured pressure trend (24).

It would be convenient to take the pressure or temperature trend in order to express the time curve of w_c; these terms, however, are not dependent on the combustion alone, but are simultaneously influenced by other parameters such as the state of aspirated air and the compression ratio. The latter influences can be eliminated, subtracting from the temperature curve in the fired engine the curve that occurs in the unfired engine under the same conditions. The difference, $T - T_0$, seen in Fig. 10, is in each point a measure of the change in internal energy of the working gas due to combustion only. This temperature difference demanded of the combustion term increases with the input fuel rate. At the beginning of combustion it increases rapidly; in the expansion phase, however, it decays too slowly.

Regardless of their scatter and the uncertainties discussed above, all time curves of the heat transfer coefficient that have been estimated from the measured surface temperature, and which have been so far published, point to the fact that the heat transfer coefficient decreases rapidly once combustion is complete. It would therefore seem better to apply the difference in pressure, $p - p_0$, whose decay with time is more rapid, as can be seen from Fig. 10. If this difference $p - p_0$ is multiplied by the cylinder volume V_s and related to the weight of charge, it takes into consideration the dependence of the input fuel rate in the same manner as the temperature difference $T - T_0$. This follows from the law of state of the gas.

If p_1 and T_1 represent the known state of the working gas related to the instantaneous cylinder volume V_1, such as perhaps in "inlet closure" or "beginning of combustion," we can write

$$w_c = C_2 \frac{V_s T_1}{p_1 V_1} (p - p_0) \qquad (7)$$

If this empirical expression is introduced according to Eq. 6a into Eq. 5b in order to calculate the total heat transferred during the cycle, the only unknown constant, C_2, in this expression can be determined by comparing the results with the measured data, as in Fig. 9. Several test series have been carried out in which the main parameters (pressure of charge, p_{ch}; temperature of charge, T_{ch}; mean piston speed, c_m; and air-fuel ratio λ) were changed over a wide range. Figs. 9 and 11 to 13 show that in all cases the measured data are adequately represented by

$$h = 110\ d^{-0.2}\ p^{0.8}\ T^{-0.53} \times \left[C_1 c_m + C_2 \frac{V_s T_1}{p_1 V_1} (p - p_0)\right]^{0.8} \qquad (8)$$

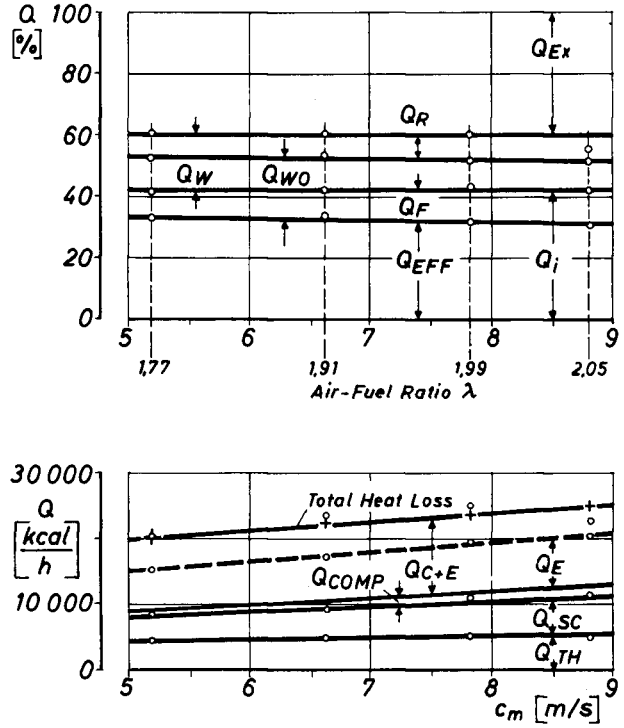

Fig. 11 - Heat balance of the fired engine ($p_{ch} \neq$ constant)

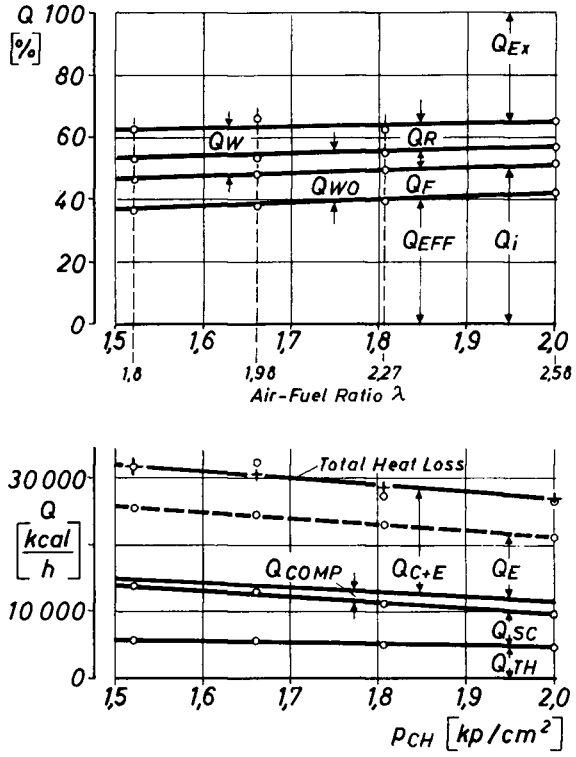

Fig. 12 - Heat balance of the fired engine ($p_{ch} \neq$ constant, $\lambda \neq$ constant)

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In this equation, $C_1 = 6.18$ must be applied to the scavenging period; $C_1 = 2.28$ to the compression and expansion strokes; and $C_2 = 3.24 \times 10^{-3}$ m/sec deg C.

In Fig. 14 the time curve of the instantaneous heat transfer coefficient has been plotted as provided by Eq. 8. When

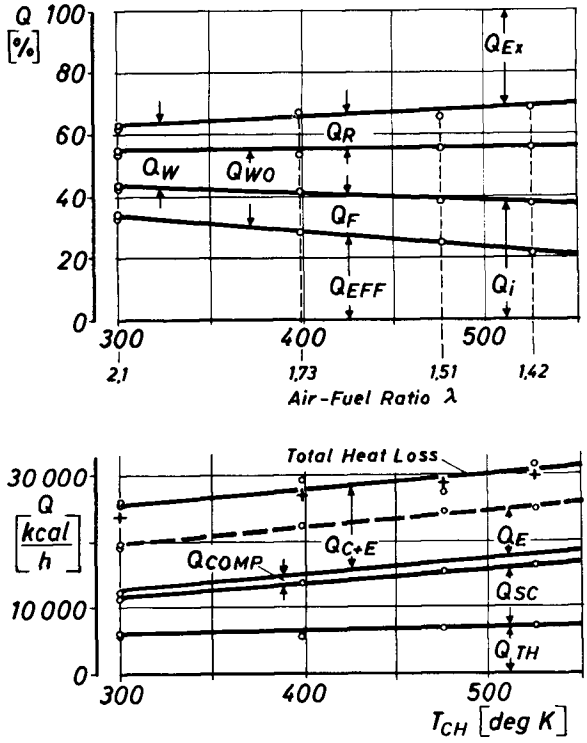

Fig. 13 - Heat balance of the fired engine ($T_{ch} \neq$ constant, $\lambda \neq$ constant)

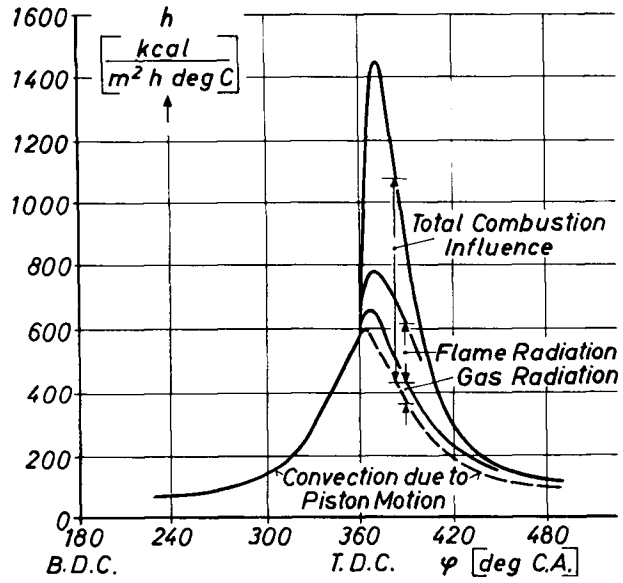

Fig. 14 - Heat transfer coefficient calculated from Eq. 8 versus crank angle

deriving the combustion terms in this equation, it was assumed that the heat transfer is caused by forced convection only. However, a considerable amount (and that is the consensus today) is transferred by radiation, obeying the law

$$Q_{ra} = \epsilon \sigma_0 A \left[\left(\frac{T_f}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] \qquad (9)$$

where ϵ is the emissivity of the radiating agent and σ_0 is the Stephan-Boltzmann constant. In diesel engines we must distinguish between flame radiation and gas radiation. The latter can be calculated fairly accurately today. The emissivity of the two radiating components of the working gas (water vapor and carbon dioxide) have been thoroughly investigated (25-27). The thickness of the gas layer is defined by

$$s = 3.4V/A$$

where V and A are the instantaneous values of cylinder volume and heat absorbing areas, respectively. When this amount of heat transferred as gas radiation is divided by the difference between the gas and surface temperatures, we find the heat transfer coefficient h_{gr} caused by gas radiation, which is also plotted in Fig. 14. This amount is rather small and can therefore be neglected in the first approximation.

Much greater importance is at present attached to flame radiation. During the combustion of hydrocarbons, soot particles are formed and these assume almost the same temperature of the gas in the reaction zone. Such a soot-particle cloud radiates like a solid body, if its concentration is high enough. The emissivity of the flame is closely connected with the complex combustion phenomena, and is dependent upon the soot concentration and the thickness of the particle cloud. Flame radiation can be calculated in a first approximation by means of Eq. 9, using the value of 0.6 for ϵ (determined on steady diesel fuel flames) and employing for the unknown flame temperature the mean gas temperature. This gives us the heat transfer coefficient h_{fr}, which is also plotted in Fig. 14.

In a similar calculation, where he employed higher flame temperatures but lower emissivity of flame values, Sitkei (5) arrived at values about the same order of magnitude. Although such an estimation must be considered with a certain reservation, it seems that the total radiation amounts to no more than about 20% of the total combustion term.

In this author's opinion, the importance of the radiation in the engine is overrated. In steady flames there are usually large combustion chambers with great volumes of flames, and because of the small velocities there are small convective heat transfer rates only, so that in these cases radiation is responsible for an appreciable amount of the total heat.

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Because of the high velocity and density of the gas, the convective heat transfer coefficients in internal combustion engines are about ten times higher, so that despite the higher temperatures the radiation plays an appreciable smaller part.

There is no doubt that such an influence occurs in the engine, and it is taken into consideration in lumped form in the combustion term as defined here, but it seems impossible to the author to separate it from the convective term. The only question that remains to be considered, therefore, is the error made with regard to the tendency, if the relationship in Eq. 8 is assumed to be valid also for the radiation term. As may be recognized from Eq. 9, radiation is dependent upon the fourth power of temperature. The flame temperature applied here, however, is not identical with the mean gas temperature; the temperature of the glowing soot particles is close to the temperature of the gas in the reaction zone, which is fairly independent of the input fuel rate because in that zone the local air-fuel ratio is always about unity. The flame temperature therefore does not change with the total air-fuel ratio, but remains constant over a wide range of the input fuel rate.

The expression for temperature, contained in Eq. 9, is therefore approximately constant, and a change in flame radiation is consequently due to a change in the emissivity of the flame. Soot concentration and flame volume determining the emissivity will increase with higher input fuel rate, and the time curve corresponds more or less to the heat release diagram. It therefore seems to be more practical to assume the radiation term as being proportional to the combustion term defined here, than to assume it as dependent upon the fourth power of the mean gas temperature.

As shown in Figs. 9 and 11 through 13, Eq. 8 gives the local average heat transfer coefficient in a satisfactory manner. It is identical with the dimensionless equation

$$Nu = 0.035\,Re^{0.8} \qquad (10)$$

where the cylinder bore "d" is taken as the characteristic length unit and where the unknown gas velocity in the cylinder,

$$w_{sc} = 6.18\,c_m$$

is applied to the scavenging period,

$$w_{comp} = 2.28\,c_m$$

is applied to the compression phase, and

$$w_{c+ex} = 2.28\,c_m + 3.24 \times 10^{-3}\,\frac{V_s T_1}{p_1 V_1}(p - p_0)$$

expresses the combustion and the expansion phase.

In accordance with their causes a distinction is made between a convective term effected by the piston motion (which is assumed proportional to the mean piston speed) and an additional term effected by the combustion itself (which is assumed proportional to the empirically estimated expression $V_s T_1 (p - p_o)/p_1 V_1$, which is a measure for the intensity of combustion.

Fig. 15 gives the time curve of the instantaneous heat transfer coefficient computed by Eq. 10 and compared with the results of some other formulas. As opposed to Eq. 8 or Eq. 10, respectively, the formulas of Nusselt, Eichelberg, and Pflaum fail to agree with the laws of similarity of convective heat transfer; they therefore have an empirical character only. The Annand formula is similar to Eq. 10. It goes back to the same considerations, but no additional combustion term dependent on the intensity of combustion only, is defined by him. All he takes into consideration is gas radiation.

By a statistical evaluation of the measured data of Elser (15), Annand provided an exponent of the Reynolds number similar to the author's. This is surprising because if one takes (as Annand also did) the mean piston speed instead of the unknown gas velocity, that is, a mean time value, the exponent can be investigated only if (as shown in this paper) parameters such as engine speed, pressure, and temperature of the intake air are changed and the total heat transferred during the entire cycle is considered. Along the crank angle, however, because of the change in the instantaneous values of the gas velocity, the condition of similarity is not fulfilled. Therefore the exponent cannot be evaluated from the measured time curve of the heat transfer coefficient.

On the other hand, if the exponent is computed as shown in this paper, it may be possible to estimate the error due to the assumption of a mean time gas velocity, by comparing the measured trend of heat transfer coefficient with

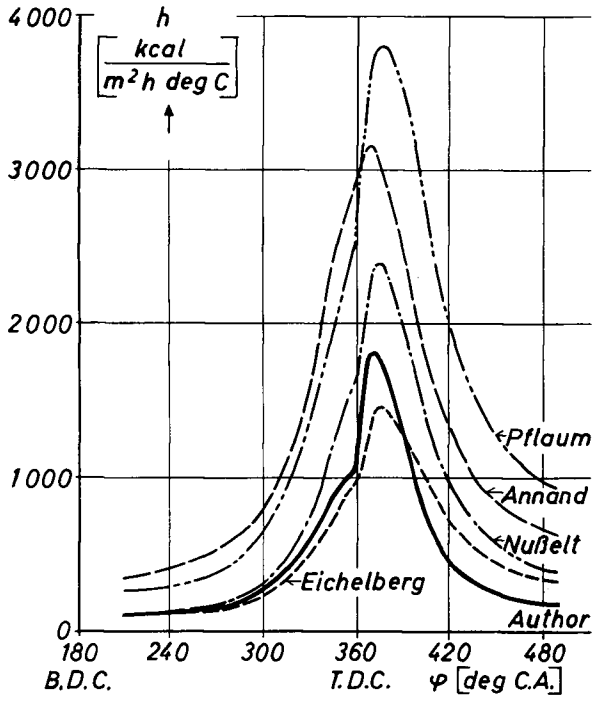

Fig. 15 - Comparison of heat transfer coefficients calculated from different formulas

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the results of the formula. But to this author it seems a hopeless task to attempt to evaluate the exponent of the Reynolds number from the measured time curve, even if it were to represent the actual mean value of the heat transfer coefficient in terms of location.

As mentioned above, because of the considerable local differences of the heat transfer coefficient, evaluations applying the surface temperature, like those of Elser, are unsuitable for determining the local mean heat transfer coefficients. This may be the reason for the high values yielded by the Annand formula.

In this author's view, the additional convection caused by combustion is the main reason for the increasing heat transfer coefficient in this phase; other influences such as radiation and change in temperature with time do not prevail, but are taken into consideration in lumped form in the equation proposed here.

Eq. 8 or Eq. 10, respectively, enable us to take into consideration the different fluid flow conditions in several engines by introducing various constants that are to be investigated experimentally. To determine a local mean heat transfer coefficient, a local mean gas velocity must be applied in the Reynolds number. It would be useless to investigate this experimentally, as it has a more or less symbolic character only. It is, however, possible to assume this velocity as being proportional to known parameters, according to its causes, such as piston motion and intensity of combustion, and to determine the proportional factors by experiments as shown in this paper.

It becomes clean, however, that a universal validity of such a formula can be expected only if different constants are introduced for different engine designs and which are valid for geometrically similar engines only. These constants must be determined in the course of time by experiments or statistical evaluation of given measured heat balances.

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APPENDIX

SYMBOLS

A, A_i = Heat absorbing areas, m^2

C, C_1 = Constants

C_2 = Constant, m/s deg K

C_m = Mean piston speed, m/s

d = Cylinder bore, m

Gr = Grashof number

h = Heat transfer coefficient, $kcal/m^2$ h deg C

n = Engine speed, rpm

Nu = Nusselt number

p = Gas pressure, kp/cm^2*

p_0 = Gas pressure in the cylinder of the corresponding motored engine, kp/cm^2

p_{Ch} = Pressure of charge air, kp/cm^2

Pr = Prandtl number

Q = Quantity of heat, kcal/h

Q_c = Input fuel heat, kcal/h

Q_{Ex} = Exhaust gas energy, kcal/h

Q_F = Friction heat, kcal/h

Q_i = Indicated work, kcal/h

Q_w = Heat transferred to the walls, kcal/h

Q_{comp} = Heat transferred during compression phase, kcal/h

Q_E = Heat transferred during expansion phase, kcal/h

Q_{sc} = Heat transferred during scavenging period, kcal/h

Q_{Th} = Heat transferred to the throat walls, kcal/h

Q_{Ra} = Heat transferred by radiation, kcal/h

Re = Reynolds number

s = Thickness of radiating gas layer, m

T = Local mean gas temperature, deg K in the cylinder

T_w, T_{wi} = Wall temperatures, deg K

T_{Ch} = Charge air temperature, deg K

U = Internal energy of the gas, kcal/kg

V = Cylinder volume, m^3

w = Local mean gas velocity, m/s

w_{sc} = Mean gas velocity in terms of time and location during scavenging period, m/s

w_{comp} = Mean gas velocity in terms of time and location during compression phase, m/s

w_c = Gas velocity caused by combustion, m/s

ϵ = Emissivity of radiating agent

λ = Air-fuel ratio**

σ_0 = Stephan-Boltzmann constant, = 4.96 $kcal/m^2$ deg k^4

ϕ = Crank angle, c.a.

τ = Time, sec

*A distinction is made between the force unit kp (kilopound) and the mass unit kg, both values being numerically equal.

**λ is defined as actual mass of air related to stoichiometric mass of air.

DISCUSSION

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THE AUTHOR is to be commended for producing this important analysis on the phenomena of heat transfer both in an engine and in a constant volume bomb.

One of the many interesting points brought out in this paper, is the variation in the instantaneous local heat transfer coefficient from the bottom to the top of the bomb.

We agree with the author that the buoyancy has an effect on the heat transfer coefficient along the surface of the bomb. However, we believe that there are some factors other than the buoyancy that might have a significant effect on the heat transfer phenomena in the boundary layer on the bomb wall. In fact, the results of the present paper, shown in Fig. 4, support this point of view. An examination of this figure shows that the maximum value for the heat transfer coefficient is at position 4. If the buoyancy is the only factor that causes the differences between the five positions, then position 5, at the top, should have the highest values, instead of position 4.

The first factor that might contribute to this variation is

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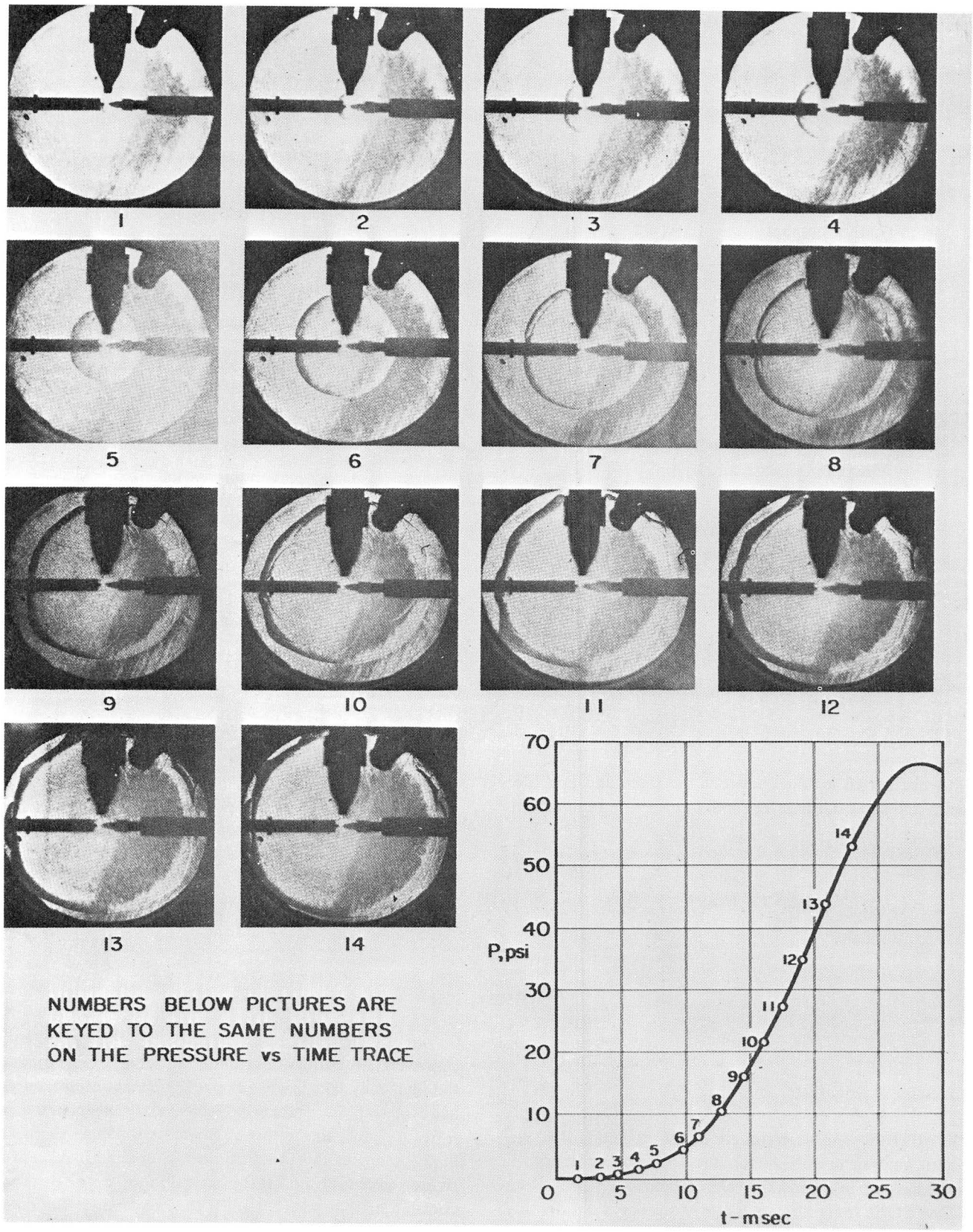

Fig. A - Sequential photographic study of combustion in a quiescent mixture - 14 consecutive runs

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the flame quenching near the wall. In a recent study made by Prof. David Cole (1)* at the University of Michigan, photographs were taken for the flame front in a constant volume bomb. The combustible mixture used was composed of propane and air in the stoichiometric ratio. Fig. A shows the flame propagation from the time of spark (frame 1) to near the end of the combustion process (frame 14).

These photographs show that the flame front at the first stages of combustion proceed in a circular form except near the walls of the probes where it is slowed down by the wall quenching effect. The buoyancy effect is not clear here because the rate of reaction of propane and air is much higher than CO and air. This high rate of reaction can be shown from the period of time between the spark and the point of maximum pressure. In the propane-air mixture this period is 27 ms, while in Dr. Woschni's experiments this period reached 130 ms, or about five times as much.

These photographs show also that as the flame approaches the walls, some protrusions occur in the flame front at positions A and B in frames 9 to 14. These protrusions are found to occur opposite to the opening in the wall where the wall quenching is not as effective as at the rest of the surface.

In the bomb used in the present paper, it is noticed that the fuel entrance to the bomb and the pressure tap are in the vicinity of position 4 where maximum heat transfer coefficient occurred.

The second factor which is believed to have contributed to the difference in the local heat transfer coefficient, is the variation in the heat transfer by free convection with the inclination of the bomb surface. To our best knowledge no work has been published on the role of free convection inside a spherical container. However, a study of the free convection in the simple case of an inclined plane might help to discuss this phenomenon.

The equation for the heat transfer coefficient on an inclined plane is given by Kreith (2) in terms of the local Nusselt number as:

$$Nu = 0.508 \left(\frac{Pr^2}{0.952 + Pr} \right)^{0.25} \left(\frac{g \cdot \beta \cdot \Delta T \cdot \cos\alpha \, x^3}{\nu^2} \right)^{0.25}$$

where:

- Pr = Prandtl number
- g = gravitational acceleration
- β = coefficient of thermal expansion
- T = temperature difference
- α = angle of inclination from the vertical line
- x = distance along the surface
- ν = kinematic viscosity

In the bomb tests the factors Pr, g, β, ΔT, and ν are the same, so it is expected that Nusselt number will change with the inclination α and distance x, or in other words, with the position of the thermocouple on the wall.

The third factor that might have contributed to the difference between the local heat transfer coefficients at the different positions in the bomb is the concentration of the fuel in the boundary layer adhering to the wall. No mention is made in the paper of the method of preparing the mixture before the start of combustion. But Fig. 2 of the paper shows that the CO was charged into the bomb from a pipe separate from that of the air, which is on the other side of the bomb. It is, therefore, expected that the concentration of CO in the boundary layer on the wall at position 4 would be higher than at the rest of the bomb. Probably the rate of diffusion of CO and air is very high, but it is believed that it will not be effective in the boundary layer adhering to the wall. This in turn plays a big role in the combustion process as the flame approaches the wall. It would be very interesting if the author would have calculated the heat transfer coefficient on the opposite side from point 4, where the concentration of CO is expected to be less.

In the analysis on the instantaneous heat transfer in engines the author assumed that the instantaneous gas velocity in the combustion chamber is in direct proportion to the mean piston speed. It is believed that using the instantaneous gas velocity is more realistic.

Fig. B shows the instantaneous gas velocity in a diesel engine (d = 4.5 in., l = 5.25 in.), together with the mean piston speed. The mean piston speed is about 12 ft/sec, while the instantaneous gas speed is in the range of 185 ft/sec, or about 15 times as much as the mean piston speed.

In a paper (3) on the instantaneous heat transfer coefficients in engines it was found that the use of the instantaneous gas velocity in Eichelberg's equation gave a more realistic value for the local heat transfer coefficient measured locally on a combustion chamber wall.

In Fig. 11 of the paper the gas pressure and temperature are given for a motored and a fired engine. It is noticed that the expansion curve in motoring is similar to the compression curve. This point has been examined carefully in our work at the University of Michigan. Fig. C shows the temperature during compression and expansion in a motored engine (d = 4.5 in., l = 4.375 in.) at 1000 rpm. The temperature at each point was calculated from the measured pressure, mass of air, and the volume and the equation of state. This figure shows that the maximum temperature occurs at about 8 deg crankangle before tdc. Also it shows that the temperatures during expansion are much lower than the corresponding temperatures during compression at the same crankangle. For example at 25 deg crankangle before tdc the temperature is 908 F while at 25 deg after tdc the temperature is 650 F. The drop in temperature between the two points is 258 F.

The constants in the formulas presented in this paper are derived from a heat balance and are based on the mean piston speed. It seems that they would give values for the heat transfer coefficients so that when integrated over the whole cycle, the total heat transfer would be given to the cooling

*Numbers in parentheses designate References at end of discussion.

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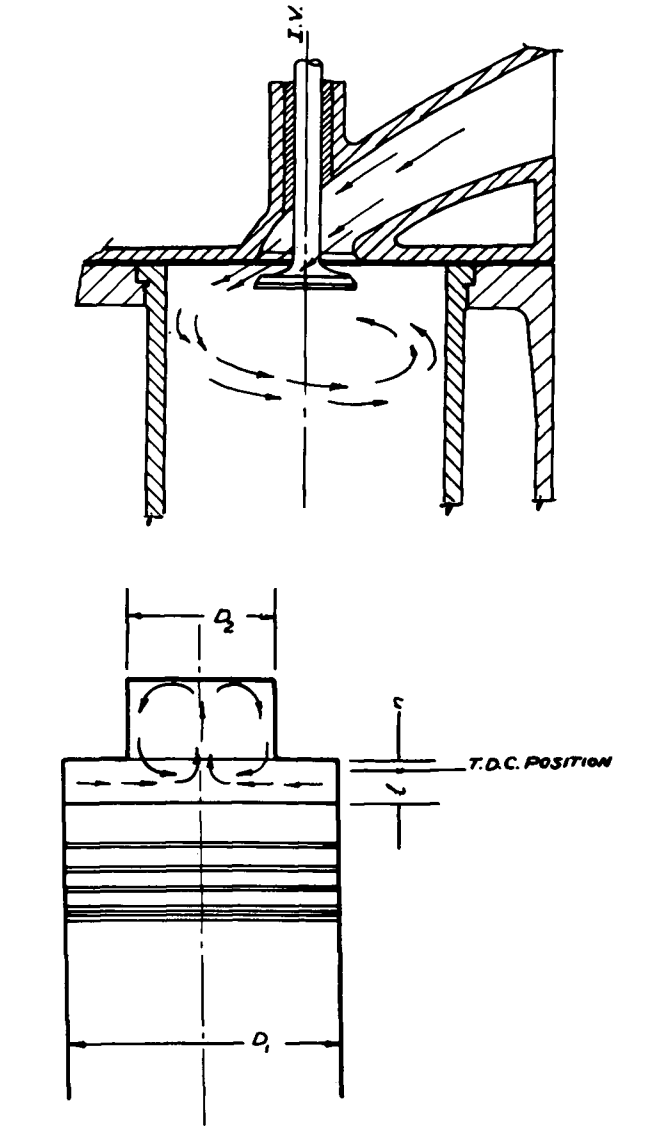

Fig. B - At top, a schematic of induction-induced swirl; middle, a schematic showing piston squish movement; bottom, a chart of gas velocity in combustion chamber due to primary turbulence

medium. Thus these formulas will be useful for design of cooling systems. But it is believed that the use of these formulas would be limited in other areas, such as thermal stress analysis in parts of the combustion chamber where the local gas velocity is much higher than the mean piston speed.

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WE WISH TO THANK Dr. Woschni for presenting this account of his research on the complex problem of diesel engine heat transfer. Any contribution to our understanding of the problem is a welcome addition.

The author describes Eq. 3 in the paper as a statement of the first law of thermodynamics. We think a more explicit description would be an energy-rate balance as:

Rate of decrease of chemical internal energy = Rate of increase of mechanical internal energy - Rate of energy transfer to the gas from the wall

By definition combustion is complete when the left-hand side of Eq. 3 equals zero. The author states that the completion

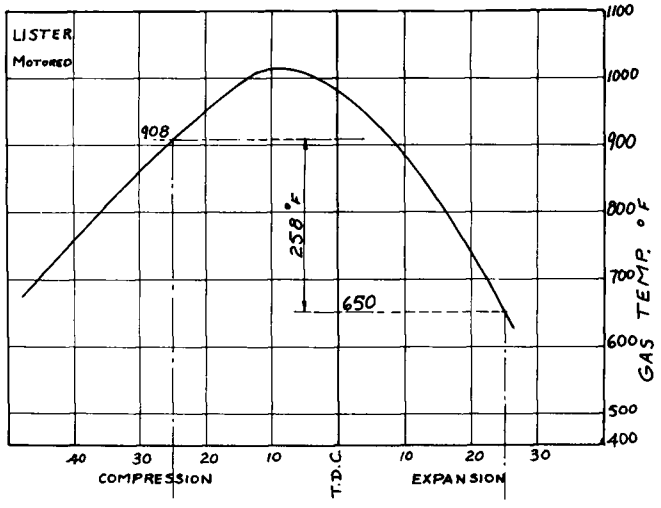

Fig. C - Gas temperature in a motored engine

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of combustion is "the moment when $dU/d\tau$ becomes zero," and indicates this in Fig. 3. This statement is incorrect since $dU/d\tau = 0$ merely indicates that the rate of energy transfer at the wall equals the rate of decrease of chemical internal energy. Furthermore, by referring to Fig. 5, one sees that $dQc/d\tau$ is non-zero, i.e., combustion is still incomplete for a considerable time, about 0.25 sec, after $dU/d\tau$ has passed through zero.

Figs. 4 and 5 of the paper show the results of some experiments conducted on the bomb assembly shown in Fig. 2. We would like to know something about the instruments used to obtain the data presented in Figs. 4 and 5. What was the design of the surface thermocouples and pressure transducer? What precautions did the author take to eliminate the usual errors (Brown 1967, Alyea 1967) in transient pressure transducers? What is the precision of data used as the basis for Figs. 4 and 5?

We believe the ordinate in the lower part of Fig. 9 should be $\frac{hT^{0.53}}{C_m^{0.8}}$ and not $\frac{hT^{0.53}}{p^{0.8}}$.

In the presentations of Figs. 10 and 12-14, the author gives values of "air-fuel ratio, λ." From the values given we suspect that λ is not the usual air-fuel ratio, but the ratio of the experimental to the chemically correct air-fuel ratio. If this is the case we feel that this should have been pointed out in the paper since it is an unusual usage of the term air-fuel ratio.

The author contends that the importance of radiant heat transfer in diesel engines is overrated. He considers a contribution from radiation of greater than 20% of the heat transfer caused by combustion to be unlikely. Fig. D summarizes the results of Ebersole* obtained in tests on an operating engine at the University of Wisconsin. The figure shows the percentage of radiant heat transfer to the total heat transfer during the engine cycle as a function of per cent indicated load. Note that near full load the radiation contributed to approximately 40% of the total heat transfer. Since the author disagrees with these results, we would like to know the basis for his disagreement with the experimental data.

*G. D. Ebersole, P. S. Myers, and O. A. Uyehara, "The Radiant and Convective Components of Diesel Engine Heat Transfer." Paper 701C presented at SAE International Summer Meeting, Montreal, June 1963.

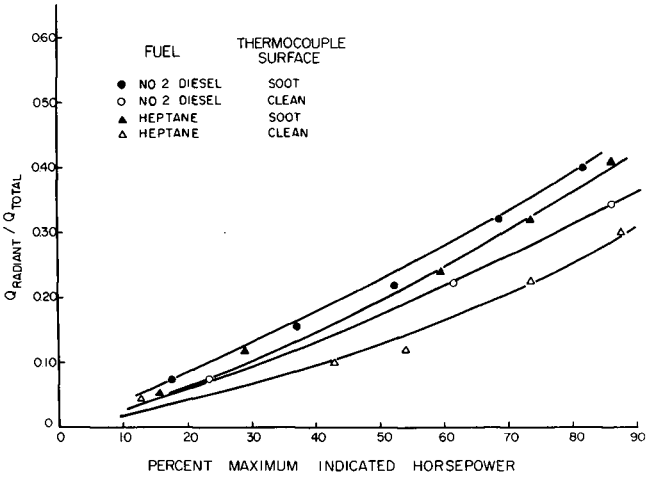

Fig. D - Heat flow ratios for diesel fuel and heptane with clean and sooty thermocouple surfaces

The experimental apparatus shown in Fig. 7 was used by the author to obtain experimental data to verify Eq. 5b. The author states that heat flux and temperature data were measured. Were these measurements done on an instantaneous basis, and if so, what instruments were used? Figs. 8 and 9 summarize the results of these tests. From the title of the paper one is led to believe that the h on the left-hand side of Eq. 5b is a function of time, or crankangle. If so, how does one interpret the data of Figs. 8 and 9? Is the h shown the time-averaged value throughout the intake and exhaust processes? If so, it is not the h needed in Eq. 1. Or is the h shown in Figs. 8 and 9 the value of the film coefficient at some particular value of crankangle? If so, the author has verified Eq. 5b for only one value of crankangle.

Several references are made in the paper to the method of obtaining instantaneous heat transfer information by measuring the variations in the cylinder local surface temperature throughout the cycle. The author contends, "This method is unsuitable for assessing the mean heat transfer coefficient in terms of location," and "Because of their considerable scatter no information can be provided on the relationship between the heat transfer coefficient and engine speed, pressure, temperature, and fuel-air ratio." We agree with the author that the results from instantaneous surface temperature measurements published thus far do show a considerable amount of scatter. However, we strongly disagree with the author in his contention that the method is unsuitable and provides no information, and we would like to ask the author how one obtains information about instantaneous heat transfer unless he measures instantaneous heat transfer--typically at a point. If the author is sincerely convinced that instantaneous heat transfer data are useless, how does he justify the use of instantaneous heat transfer data as the basis for developing Eq. 7? The discussers are convinced that instantaneous heat transfer data are a necessary building block for any heat transfer correlation.

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DR. WOSCHNI'S PAPER is very descriptive of a most ingenious technique for investigating the mean heat transfer from the gases of internal combustion engines.

I am, however, not completely clear as to one or two points, and I should be grateful if Dr. Woschni could supply a few more details.

My particular question is with regard to what is described as the scavenging period which I assume covers both inlet and exhaust. I would have thought that during the exhaust period and in particular the blow-down portion, the physical

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conditions between the unfired and the fired engine were very different. Presumably the sonic velocity during the first portion of the blow-down can be allowed for in the heat transfer calculations, but surely the heat transfer will also be affected by the complex wave and wave reflection pattern that will exist in the exhaust bend, especially under high power conditions. These will not show up under the steady state tests mentioned by the author.

I am not sure how the energy in the exhaust is measured, but if it is downstream of the exhaust passages in the cylinder head, this change could have a relatively large effect on the calculations. In this connection it is not always recognized that a large proportion of the heat transfer to coolant can take place in the exhaust passages.

We have carried out experimental investigations on diesel engines where we have measured the local heat flows through the cylinder liner, the cylinder head, and the exhaust bends of single cylinder engines fitted with large numbers of traversing thermocouples. The thermocouples enable the temperature gradient through the metal to be measured, and by ascertaining the thermal conductivity, the heat flow can be calculated. With some 25 such couples an accurate map of the local heat flows can be built up. In this way the gross heat losses through the walls can be calculated.

Typically, for an indirect injection engine, the heat to the exhaust port is somewhat over 20% of the heat to the coolant, and it is estimated at just over 30% for a direct injection engine.

From the author's Fig. 10, the heat loss to the "throats" during the scavenging period is only about 20% for a direct injection engine, and my question, therefore, is whether Dr. Woschni is sure that the difference between the calculated and measured values may not be due, at least in part, to underestimation of the heat loss during exhaust?

On the other hand, it would be surprising if radiation and other combustion induced phenomena did not result in an increase in the heat transfer coefficients as defined in the paper. This would certainly be so in an indirect injection engine where the increased heat losses, as compared with a direct injection engine, must be due to combustion induced convection.

In this connection as my colleague Mr. Alcock has pointed out, other considerations may affect the heat transfer during the actual combustion process. Combustion photography has shown that during combustion, particles of flame move around the cylinder, and hence an important part of the heat transfer during this process may be by convection from flames at temperatures corresponding roughly to stoichiometric mixture, but in contact with the surface for finite times, the heat flow varying with the contact time of the flame as well as with some function of the gas pressure and local velocity. Under these conditions any calculations based on the normal concept of mean effective gas temperature would give inaccurate results, and for example, any effect of metal temperature on heat flow will be small due to the high flame temperature.

AUTHOR'S CLOSURE TO DISCUSSION

THE AUTHOR wishes to thank the discussers for their interest in his paper.

The experiments on the spherical bomb, described in full in Ref. 1*, are only mentioned in this paper to show that the Nusselt equation, on which the Eichelberg and Pflaum formulas are based, does not express the heat transfer during combustion but only interprets the heat transfer in spherical vessels under free convective conditions. For this reason these equations for internal combustion engines where forced turbulent fluid flow is involved are not substantiated theoretically. The local heat transfer coefficients shown in Fig. 3 of the paper are obtained from the surface temperatures measured by means of one thermocouple only, whose position has been varied by turning the bomb.

Eq. 8 proposed here, as well as the formulas shown in Table 1, provides a time variable local mean heat transfer coefficient between the working fluid and the cylinder walls, which, for example, is necessary as a boundary condition for electronic cycle simulation. In internal combustion engines, of course, considerable local differences occur, and for interpreting these local values, it might be better to apply the actual gas velocity if it is known in the particular location under discussion. Such a formula, however, cannot be universally applicable. In this author's opinion the Eichelberg formula should not be applied because a formula which is theoretically not substantiated will not be better if, instead of the mean piston speed, a gas velocity is employed which is more true for this location.

The determination of local and local mean heat transfer coefficients are two different problems. There is the case of steady problems, such as the heat transfer to a tube bundle. If the total heat transferred to the bundle is to be calculated, a local mean heat transfer coefficient is applied without regard to the actual local values in the various locations.

Eq. 8 represents the local mean heat transfer coefficient in the cylinder. Therefore, it seems to this author rather useless to try to determine the exponents in Eq. 8 from measured transient surface temperatures which provide only local values, strongly dependent on chance. To determine the constants and exponents, the heat balance must be evaluated Although this method provides only the total heat transferred during the entire cycle, the influence of the basic parameters, such as engine speed, including gas velocity, gas temperature, pressure, and fuel rate input, can be determined step by step as shown in this paper. Eqs. 5b and 8 provide the instantaneous values; in Figs. 7 and 8, however, to show clearly the measured tendencies, time average values calculated as

$$h_m = \frac{\int h\,(T-T_w)\,d\varphi}{\int (T-T_w)\,d\varphi}$$

*Numbers in parentheses designate References at end of discussion.

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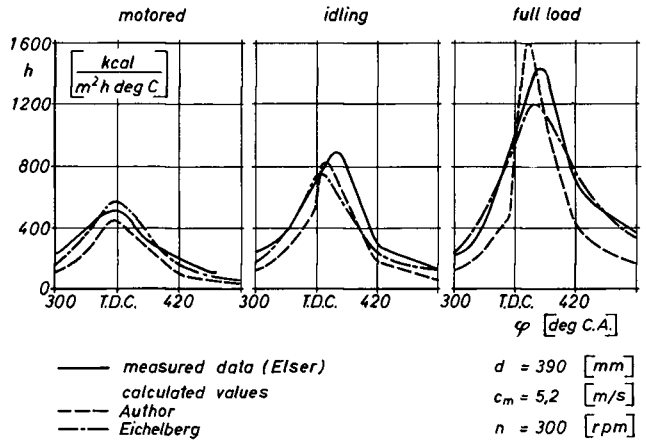

Fig. E - Comparison of measured and calculated heat transfer coefficients

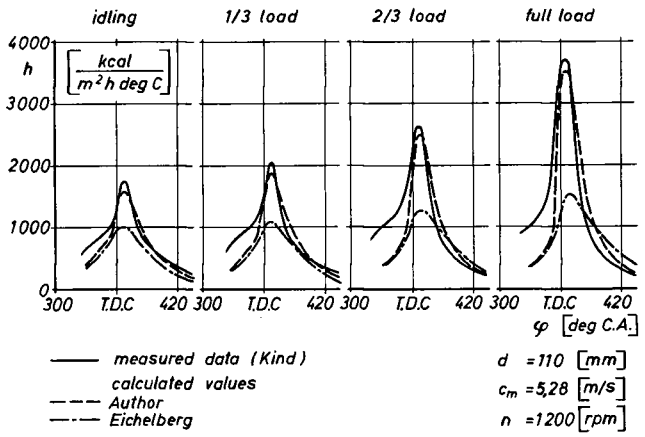

Fig. F - Comparison of measured and calculated heat transfer coefficients

are plotted. If one parameter, such as engine speed, charge pressure, or temperature is varied, then instantaneous as well as time average values of heat transfer coefficient must have the same tendency.

As shown in Figs. 9 and 11-13 of the paper, Eq. 8, when applied to calculating the total heat transferred during the entire cycle, suitably provides the measured values dependent on the different engine operating conditions. The question is now how this equation represents the actual time of the heat transfer coefficient. To check this, measured time curves determined from measured transient surface temperatures are useful if a number of thermocouples in different locations are applied with a view to obtaining local mean values. Such measurements have been carried out by Elser on a large four-stroke and by Kind on a small two-stroke diesel engine with direct injection. The results are shown in Figs. E and F. If by means of Eq. 8 the time curve of the heat transfer coefficient is calculated from the known pressure and temperature trend, the calculated maximum values for all operating conditions are in good agreement with the measured data.

Although the additional combustion term defined in Eq. 8 was derived taking convection alone into consideration, it contains radiation and other influences too, but in this author's view they do not prevail. The test results from Ebersole, although found by applying a most ingenious technique, must be considered reservedly. Ebersole used a small engine with which at full load the maximum values of the local mean heat transfer coefficient are, as may be seen from Fig. F, approximately 3600 kcal/m^{2}h^o. From Ebersole's results it can be seen that about 40% of the total heat transferred during the entire cycle is caused by radiation. Since gas radiation is known to be small, most of it must be transferred during the combustion phase by flame radiation. In this phase about 60% of the total heat must be transferred by radiation, that is, the heat transfer coefficient caused by radiation should be about 2000 kcal/m^{2}h^o. This latter value can be calculated from

$$h_{Ra} = \epsilon\sigma_o \frac{\left[\left(\frac{T_f}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right]}{T - T_w}$$

The flame temperature was measured by Uyehara as being around 2200 K (2). If the gas temperature is assumed to be 1800 K and the wall temperature as 500 K and the flame is assumed as being a black body one obtains a maximum value of h_{Ra} of about 900 kcal/m^{2}h^o C. That means, such a high amount of radiation as measured by Ebersole is physically not plausible, the less, since the emissivity of diesel fuel flames is not 1 but rather about 0.2 (3).

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